DESIGN OF FLOW CONTROL DEVICE FOR DYNAMIC COLD PLATE AND OPTIMIZATION OF SERVERS TO DECREASE JUNCTION TEMPERATURE OF HEAT SINKS

by

KUNAL ATULKUMAR SHAH

THESIS

Presented to the Faculty of the Graduate School of

The University of Texas at Arlington

in partial fulfillment of the requirements

for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

at

THE UNIVERSITY OF TEXAS AT ARLINGTON

December 2016

Arlington, Texas

Supervising Committee:

Dr. Dereje Agonafer, Supervising Professor

Dr. Ratan Kumar

Dr. Haji Sheikh

Copyright © by

Kunal Atulkumar Shah

2016

All Rights Reserved

DEDICATION

Dedicated to the memory of my mother Deval Shah.

You left fingerprints of grace on my life.

I dedicate my thesis work to my loving parents Atul and Deval Shah. I am grateful for their boundless sacrifices, devotion and words of encouragement that pushed me to achieve my dreams. My brother Monil has been very special for being on my side at all times. All 3 of you have been my best cheerers. To my grandparents, for inspire me to pursue bigger challenges in life.

I also dedicate this thesis to my family and friends for believing in me and supporting me emotionally when I needed the most. Their selfless care and well wishes will always be appreciated.

ACKNOWLEDGEMENTS

I am grateful to my supervising professor Dr. Dereje Agonafer for his support for my research work. The motivation, support and the platform provided by him were the key factors for this achievement. The freedom to use my creativity on the topic of my own choosing allowed me to steer in the direction of my own choosing. This approach has significantly helped me grow as a professional.

I would also like to thank Dr. Raul Fernandez and Dr. Robert woods to spend their valuable time and share their expertise and remarks. I would like to thank Dr. Ratan Kumar and Dr. Haji Sheikh, my thesis committee members for their valuable inputs and suggestions. I am thankful to Sally Thompson and Debi Barton for relieving my anxiety when I used to get stuck with the educational and administrative matters. I am thankful to Rajesh Kasukurthy, Shivang Desai and my other EMNSPC team mates for their support.

Abstract

DESIGN OF FLOW CONTROL DEVICE FOR DYNAMIC COLD PLATE AND OPTIMIZATION OF SERVERS TO DECREASE JUNCTION TEMPERATURE OF HEAT SINKS

Kunal Atulkumar Shah, M. Sc. in Mechanical Engineering

The University of Texas at Arlington, 2016

Supervising Professor: Dr. Dereje Agonafer

Electronics cooling research is facing the challenges of high heat flux removal and increased pumping power. Not much attention has been focused related to server and module level cooling. The continued increase in heat fluxes at the chip level due to new and robust technology nodes following Moore's law is starting to push the limits of air cooling and especially for high end servers. An alternative to air cooling is liquid (water and oil immersed) cooling.

The optimized dynamic cold plates that use water as the cooling fluid have shown over 28 % and 52% of reduction in pumping power and average temperatures across the module respectively. In all dynamic cold plates, flow control devices have to be used. Our first generation state of the art

flow control devices required dampers, actuators, sensors, transducer and control module. This approach made the system complex and reduced its reliability as it required integration of N number of elements. Hence a new method to overcome this challenge is proposed. A passive system is designed for temperature control. Various materials are shortlisted and design parameters to have a reliability for 10^7 cyclic loading are considered.

For oil cooling 3rd generation Open Compute servers have been utilized. Optimization of heat sinks, server orientations and channels are done to decrease the junction temperature for oil-immersed servers. The resultant effect of using mineral oil and these optimizations is that the high-end servers can have much higher heat flux density nevertheless maintain its junction temperature under acceptable limits.

TABLE OF CONTENTS

DESIGN OF FLOW CONTROL DEVICE FOR DYNAMIC COLD PLATE AND OPTIMIZATION OF SERVERS TO DECREASE JUNCTION TEMPERATURE OF HEAT SINKS i
DEDICATION iii
ACKNOWLEDGEMENTS iv
Abstract v
Nomenclature 7
Chapter 1 Motivation for Optimizing Datacenter Technology 8
Chapter 2 Thesis Approach 10
2.1 Design of Flow Control Device for Dynamic Cold Plates 10
2.2 Optimization of Servers to Decrease Junction Temperature of Heat Sinks 12
Chapter 3 Introduction to Design of Flow Control Device for Dynamic Cold Plates 14
3.1 The concept of dynamic cold plates 14
3.2 Existing Design of Flow Control Devices 17
Chapter 4 Need for a New Flow Control Device 18
4.1 Power Calculation 18
4.2 Passive vs Active Control 19
4.3 Deflection Requirements 20
Chapter 5 Design Approach for a New FCD 21
5.1 Shape Memory Materials Vs Bimetallic Materials 21
5.2 Nitinol Vs Other Materials 22
5.3 Nitinol (FLEXINOL®) Alloy Physical Properties [13] 24
5.4 Mass of FCD Damper 25
5.5 Reliability and Operating Range for Nitinol 26
Chapter 6 Flow Control Device Design Optimization 29
6.1 Test Results of the Temperature Sensitive Shape Changing Springs 29
6.2 Optimized Design 30
6.3 Variations to the Design 33
6.4 Advantages of the New Design 34

6.5 Extendibility to Other Applications 35
Chapter 7 Optimization of Servers to Decrease Junction Temperature of Heat Sinks 36
7.1 Importance of Oil Immersion Cooling 36
7.2 Why Mineral Oil ? 36
7.3 Server Selection 37
7.4 Fluid Conditions 38
7.5 Mineral oil and Server Material Properties 39
7.6 Server Geometry 40
Chapter 8 Server Design Optimization Approach 41
8.1 Heat Sink Optimization 41
8.2 Duct Design Optimization 42
8.3 Server Orientation and Size Optimization 43
Chapter 9 Optimized Results 44
LIST OF FIGURES 46
LIST OF TABLES 48
LIST OF REFERENCES 49

Nomenclature

C_p = Thermal Capacity [kJ/K]

M = Mass [kg]

m = Mass flow rate [kg/s]

q = heat load [kW]

T = Temperature [K]

T_i = Temperature at Inlet of Cold plate

T_o = Temperature Outlet of Cold Plate

T_c = Temperature of Nitinol at Cold State (Room Temperature)

T_h = Temperature of Nitinol at Hot State

L_c = Length of nitinol when in cold state (mm)

L_h = Length of nitinol when in hot state (mm)

OCP = Original Cold Plate / Static Cold Plate

DCP = Dynamic Cold Plate

FCD = Flow Control Device

PUE's = Power Usage Effectiveness

Chapter 1 Motivation for Optimizing Datacenter Technology

Datacenters consumes approximately 2% of the total energy consumed in USA. This is a multibillion dollar industry with a compound annual growth rate of 9.3% [4]. In past 3 years, by doing optimizations on its datacenter design Facebook alone has saved over $2 billion [5]. Hence, all the companies that are heavily invested in this arena of engineering are continuously trying to find alternatives to improve the power usage effectiveness to reduce the operating costs. Below is a list that shows the heat removal capacities of various cooling mediums.

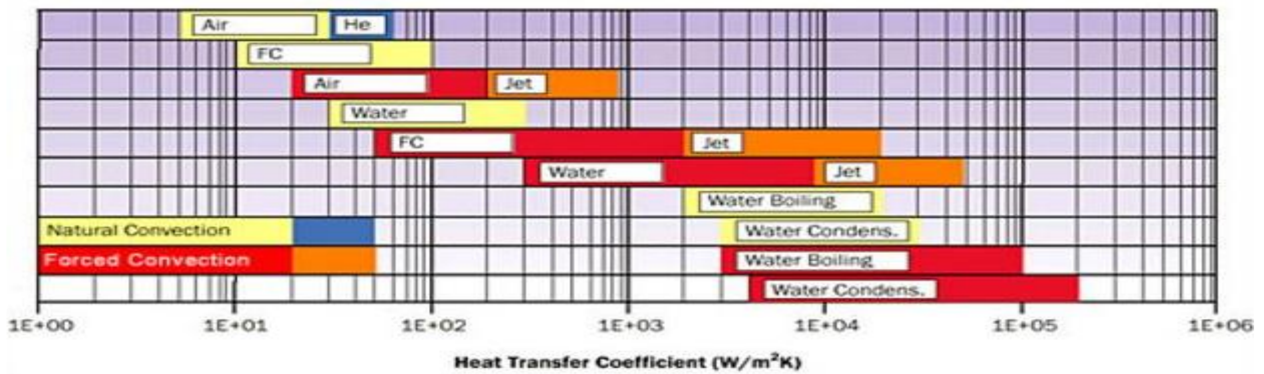


Figure 1: Order of magnitude for heat transfer coefficients depending on cooling technology [6].

Fluid Medium	Heat Capacity (kJ/kgK)	Conductivity (W/mK)	Thermal Mass (KJ/m^3k)	Kinematic Viscosity (mm^2 /s)
Air	1.01	0.02	1.237	0.16
Dielectric Mineral Oil	1.67	0.13	1417.83	16.02
Water	4.19	0.58	4186	0.66

Table 1: Characteristics of the coolant fluid

Most of the air cooled datacenters have power usage effectiveness (PUE) of around 1.8 to 1.5. In order to improve PUE we are incorporating water based cooling as it promises much better PUE because of its greater thermal mass.

Moreover, liquid based data centers have many more advantages compared to air based cooling systems. They have higher mean time to failure (MTTF) as the amplitude of temperature fluctuations is very less, silent operation because of removal of fans, and clean environment as there is no air based dust deposition. This results in better reliability and superior ergonomics for employees.

Chapter 2 Thesis Approach

The purpose of this research work is to develop the cooling performance of data centers. Efforts are made to optimize datacenters at server level and at chip scale package level. High-density, non-uniform heat loads that has high power demands will become common in near future. Hence, attention is drawn towards liquid cooling methodologies of direct and indirect cooling that have high heat carrying capacities with lower PUE's.

There are two diverse areas that are explored in this thesis. First area of focus on improving the performance of cold plates by making them dynamic and using passive flow control devices. Second area of focus is to optimize the performance of 3rd generation of Open Compute servers.

2.1 Design of Flow Control Device for Dynamic Cold Plates

Electronics cooling research is facing the challenges of high heat flux removal and increased pumping power. Lots of work has been done to improve the cooling efficiency of data centers at the rack, room and facility level. Conversely, not as much attention has been focused towards module level cooling. The continued increase in heat fluxes at the chip level due to new and robust technology nodes following Moore's law is starting to push the limits of air cooling and especially for high end servers.

An alternative to air cooling is liquid cooling. It is a technology that has been used since the early 80's by IBM and at the time using indirect liquid cooling. With the introduction of multi-core chips and corresponding highly non-uniform power distribution, the traditional static cold plate could result in large temperature gradients across the die. As such, there arises a need to have a dynamic cold plate which distributes the coolant as per the power distribution of the die and hence

potentially decrease the temperature gradient across the die. In the test setup, the static cold plates were replaced by dynamic cold plates.

All cold plates have static design and are incapable to effectively regulate the flow pattern as per the changing, non-uniform IT loads. Hence, we incorporated Dynamic Cold Plates that have distributed cooling system within the cold plates whose flow is managed by introducing control systems [1][2].

Numerical Analysis was done to validate the experimental work and to optimize the performance limiting parameters of the dynamic cold plate. It was shown that a significant reduction was accomplished by using the dynamic cold plate. Furthermore, optimizations of the parallel plate fin in the dynamic cold plate indicated significant improvements in pumping power performance. The optimized dynamic cold plates have shown over 28 % and 52% of reduction in pumping power and average temperatures across the module respectively [3].

In all dynamic cold plates, flow control devices have to be used. Most of the flow control devices require dampers, actuators, sensors, transducer and control module. This approach makes the system complex and reduces its reliability because it requires integration of N number of elements. Moreover, it requires downtime and extra spares for inspection and maintenance. In this research we propose to use new and robust method to overcome this challenge.

In our attempt to propose an optimal flow control device, various materials that are sensitive to temperature, super elastic and have shape memory properties in bi-metallic, alloy, polymeric and composite domains are shortlisted. Selection Parameters of damper size, hydraulic diameter for the full load condition and minimum load condition, power consumption for various geometries and fluid mediums, external heater placement, active and passive flow control schemes, different

geometries, material required to overcome the hydraulic pressure and the location of flow control device are investigated. Moreover, even the design parameters to have reliability for 10^7 cyclic loading is also considered.

2.2 Optimization of Servers to Decrease Junction Temperature of Heat Sinks

Another approach to increase efficiency is to customize servers for use oil immersed cooling strategies for datacenter. With the aggressive implementation of Internet of Things and Cloud Computing, hardware developers and cooling solution providers face a constant challenge. The approach is to replace forced convective air cooling method at the server level with a forced convective low velocity non-conductive mineral oil cooling that has high heat carrying capacity. Oil immersion seems to be a promising alternative to mitigate the challenge of increasing junction temperatures when the heat flux density increases exponentially. This is required even more when one heat generating element is placed in the shadow region of another to reduce the server′s foot print.

In this study, 3rd generation Open Compute Server that has a similar configuration is considered. Form factor is kept constant and the baseline model is compared with the geometrically optimized model for oil cooling. These models are compared at constant mass flow rate. As the oil cooling system does not require fan for displacing the media the fan components are removed and the height of server is reduced considerably. This helps in stacking more servers within the same space. The heat sinks geometries on both the chips are optimized to reduce the overall junction temperature. Ducting is incorporated over the RAM′s to guide majority of the flow over the heat sinks. Moreover, different server orientations are studied to see the effect of gravity on thermal shadowing, junction temperatures. All the Numerical Analysis are conducted using ANSYS Icepak

software. The optimizations are conducted by using parameterization tools. The resultant effect of using mineral oil and these optimizations is that the high-end servers can have much higher heat flux density nevertheless maintain its junction temperature under acceptable limits.

Chapter 3 Introduction to Design of Flow Control Device for Dynamic Cold Plates

3.1 The concept of dynamic cold plates

Water is a very effective cooling medium as it is a very good conductor of heat. But it also conducts electricity and hence cannot be in a direct contact with the electronic components. Hence warm water cooling by using cold plates is used to take the advantages of the cooling properties of water.

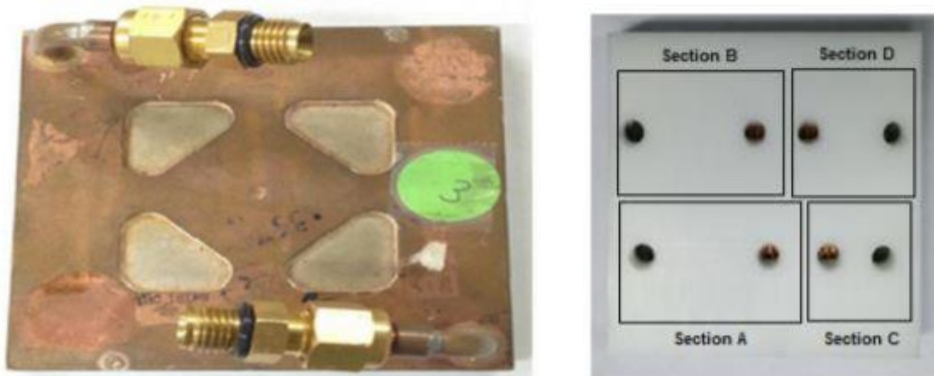
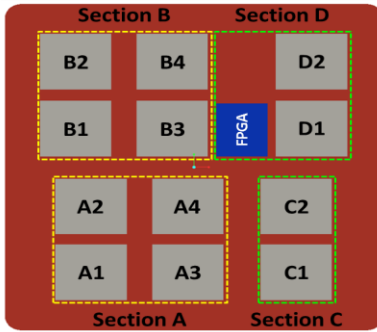


Figure 2: Top view of Original Cold Plate [7]

Figure 3: Top view of Dynamic Cold Plate [7]

Cold plate is similar to a passive heat sink. It is attached on top of the heat generating multi-chip module. The chips are in contact with the cold plate with the help of a porous thermal interface materials as heat spreaders. The bottom portion of cold plate is made up of copper. This helps in taking away the heat from the chips. The cold plates have inlets and outlets at its top for water from the facility. This cold water (30˚C / 35˚C) comes in contact with the bottom portion and the heat sink fins of the cold plate and carries away the heat from the servers. The hot water (40˚C / 45˚C) goes to the external facility level heat exchangers to cool down and the subsequent cycles repeat.



Component	Quantity	Power (W)
ASIC	1	40
ASIC	12	5
FPGA	5	5

Figure 4: Multi Chip Module for the Cold Plate [7]

Figure 5:Power Rating of Chips [7]

The original static cold plate had 1 inlet and 1 outlet whereas, dynamic cold plate is divided into 4 sections and each section has its own inlet and outlet (see fig.). All the sections have different power distribution and hence necessitates compensating flow to keep it under allowable temperatures. The dynamic cold plate is designed to let the flow in one section of the cold plate be independent from the flow in other sections. In the experimental model the ASIC chip B3 is power at 40W and all other ASIC and FPGA chips are powered at 5W each. The inlets are near the center of the dynamic cold plate and the outlets are near the perimeter of the cold plate.

Comparison of Original Cold Plate and Dynamic Cold Plate Results [3][7]

Original cold plate(OCP)

Pumping power of 3.85W at flow rate of 4lpm

Δ T = 12.87°

Dynamic cold plate(DCP)

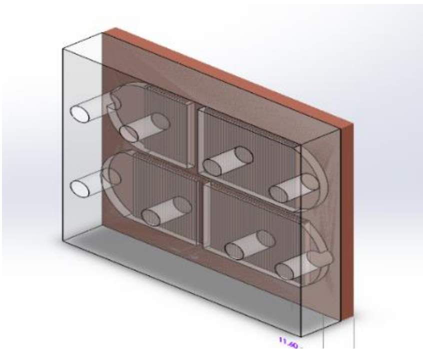
Pumping power of 2.76W at flow rate of 4lpm

Δ T = 6.13°

Total Savings on the Dynamic Cold Plate design

Pumping Power Saving = 28.32%

Δ T savings = 52.36%



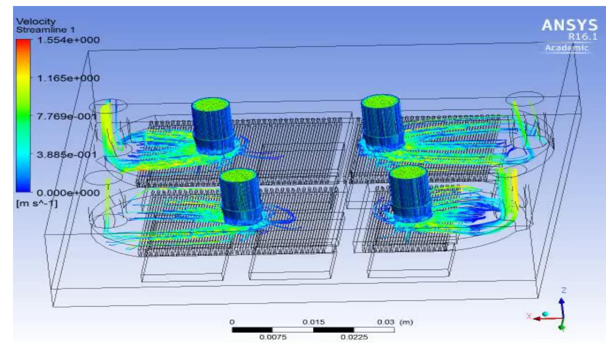


Figure 6 (left): CAD model of the Dynamic Cold Plate [7]

Figure 7 (right): CFD model of the Dynamic Cold Plate [7]

As we see over here by having distributed cooing method we are able to make the systems energy efficient. Moreover, the reliability of the chip scale package increases as the range of thermal loading decreases. Hence, distributed cooling is better. In order to have distributed flow across the multichip module we need a device / method to control the flow across all the parallel channels.

Moreover, static cold plates have limits to its dimensions for a particular heat flux density. If it has been designed for 10cm*10cm, it cannot be extended to 20cm*20cm. to do so it needs to be completely redesigned because of heat rejection issues. Dynamic cold plates don't have this issue. The sections can be increased from 4 (scope of this project) to 8, 16, 32 etc. All one needs to do is increase the no of cold plate partitions and flow control devices attached with them. Hence, the size of the DCP can even go above 10cm*80cm with 32 sections.

3.2 Existing Design of Flow Control Devices

Active Control

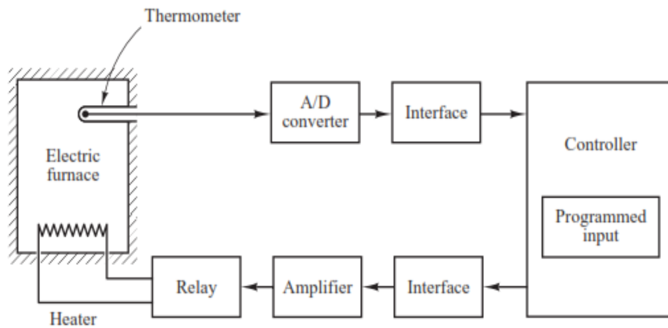
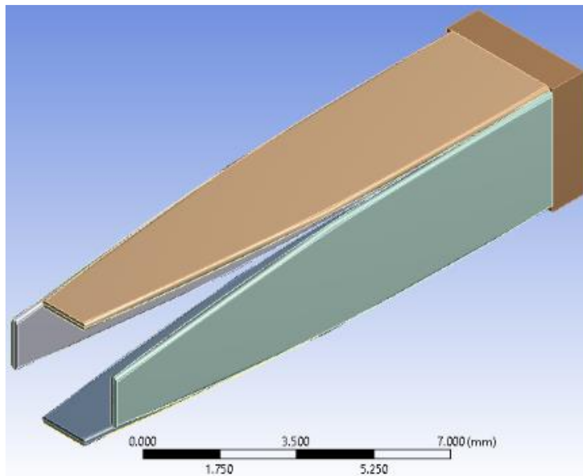


Figure 8 (left): Modern Control Engineering [8]

Figure 9 (right): – Active flow control [9]

The figure showcases the block diagram of a typical active flow control device based on thermal inputs. It has several components to it which need to be integrated.

Passive Control



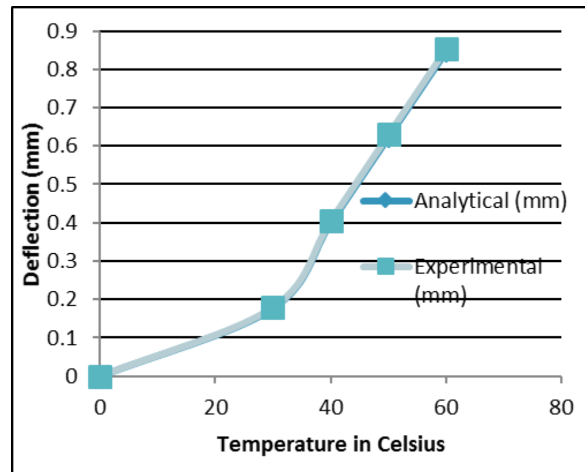


Figure 10 (left) – Passive flow control by using banana leaf bimetallic strip concept [10].

Figure 11 (right): Deflection vs temperature graph for the bimetallic strip model [10].

A simplified passive bimetallic device which activates (deflects / bends) as there is change in temperature.

Chapter 4 Need for a New Flow Control Device

4.1 Power Calculation

As per the experimental results we need to raise the temperature above 70 degrees to deflect the bi-metallic banana leaf cantilever mechanism by mere 1 mm. The above bimetallic device was simulated in ANSYS ICEPAK to see what amount of power is required to raise the temperature of the damper. These were the results obtained for change in temperature of 35˚C.

Power Calculations as per ANSYS ICEPAK Conditions: FCD Material: Nickel, Velocity:0.2m/s, Conduit Size:5x5, Fluid Inlet Temperature: 35 (P=VI, V=IR, R is dependent on dimensions)							
Medium of operation	Geometry Considered	Dimentions of material (mm*mm*mm)	Power supplied to 1 FCD (W)	Mean source temperature achieved (C)	Mean Coating temperature achieved (C)	Mean temperature change of the fluid (To-Ti) (C)	Flow Rate m/s
Water	Plate	20x3x1	25	75.9	NA	2.41	0.2
	Thin Plate	20x3x0.2	20	78.29	NA	1.7	0.2
	Thick Wire	20mm length and 0.20 thickness	4	70.17	NA	0.25	0.2
	Thin Wire	20mm length and 0.020 thickness	1.2	45.5	NA	0.08	0.2
	Thin Wire	20mm length and 0.025 thickness	1	72.89	NA	0.07	0.2
	plate coated with Polymer	20x3x0.2 + 0.3 polymer thickness	2.3	73.4	56.1	0.35	0.2
	Thin Wire coated with polymer	20mm length and 0.025 thickness with 0.03 polymer thickness	0.4	70	55.3	0.02	0.2
Air	Plate	20x3x1	0.1	72.6	NA	22.87	0.2
	Thin Plate	20x3x0.2	0.06	82.1	NA	9.92	0.2
	Thick Wire	20mm length and 0.20 thickness	0.04	72	NA	14.24	0.2
	Thin Wire	20mm length and 0.025 thickness	0.02	71.88	NA	7.05	0.2
	Thick Wire coated with Polymer	20mm length and 0.20 thickness	0.06	74	NA	10.4	0.2

Table 2: Power calculation to raise the FCD temperature to 70˚C Celsius

The FCD used 0.4*4= 1.6W for each DCP in water. The saving incorporating by using a DCP concept is 1.2 W and the amount of energy used to activate is 1.6W. Hence, using this design of FCD was turning out to be counterproductive. Therefore, the deflecting material should be in air if active control is desired as the power consumption is only 0.02*4=0.08W per DCP. Moreover, having a FCD outside in air will need a bigger form factor and hence would create a packaging issue. It will not allow dense packing of serves and enough server drawers can't be loaded at each server rack. Hence, it is more desirable to have the FCD in water.

4.2 Passive vs Active Control

A more uniform temperature distribution and corresponding reduction in external thermal resistance is achieved by using DCP. Most of the existing flow devices require complex arrangement and integration of elements like temperature sensing probe, transducers, preprogrammed controllers, solenoids, mechanical actuators and structural dampers. Such an approach makes the system complex, expensive and unnecessarily reduces its reliability. These mechanical systems have a tendency to deteriorate over time, due to wear and tear which changes their sensitivity. Hence it requires maintenance downtime and increases cost of spares. Also the existing metallic (or bimetallic) flow control devices have the issues of functioning in selective fluid mediums only considering the issues related to chemical degradation over time (e.g.: corrosion) and increasing the contamination within the systems. Moreover, simple bimetallic flow control devices have one element which senses and actuates. To get the desired deflection they are needed to be heated to higher temperatures with the help of external heaters / current. Hence, it has an intrinsic fault of giving faulty subsequent readings and actuations after the first reading. If the signal amplifications at the controller are high this fault amplifies and even more so as the metallic strip rejects the heat into the closed loop of coolant channel. Water not only needs to take the heat load of ASIC and FPGA chips but also the heat load generated at the FCD. As a result, it induces even more cooling load on the main heat exchanger which furthermore reduces the efficiency of the system.

1 cold plate needs 4 Flow Control Devices, 1 Server will have 8 cold plates, i.e. 4*8 = 32 Flow control Devices per server, 1 Datacenter Rack will have 16 to 22 Servers stacked = 32*16 = 512 Flow Control Devices per rack.

Now to monitor, inspect and maintain these many flow control devices for each rack is a very costly and time consuming task. Moreover, these devices have dimensions smaller than **100*3*1 mm^3.**

Hence, we plan to use passive mechanism to manage this system and hence eliminate all these flaws.

4.3 Deflection Requirements

To design a passive FCD various parameters have to be considered. Parameters like the total deflection required to completely open and close the valve as per the changing IT loads. To get these values hydraulic diameters is calculated.

Hydraulic Diameter and Pressure Drop calculations via Analytical Method							
IT Power to be dissipated	Delta T Across the System	Flow Rate	Velocity	Diameter	Pressure Drop at the FCD (Considering sudden contraction for worst case scenario)	Pressure Drop in inH₂O (Cold Plate Pressure Drop=6.59058, Total pressure drop = Cold Plate + FCD Nozzle)	Reynolds Number (laminar<4000<transition<6000<turbulant)
W	T	m3/s	m/s	mm	in H2O	inH₂O	Re is Dimentionless
10	12	1.98413E-07	0.2	1.123666	-0.08	-6.67058	342.675244
25	12	4.96032E-07	0.2	1.776673	-0.07	-6.66058	541.8171344
40	12	7.93651E-07	0.2	2.247333	-0.06	-6.65058	685.3504881
55	12	1.09127E-06	0.2	2.635231	-0.04822	-6.6388	803.6446825
60	12	1.19048E-06	0.2	2.752409	-0.048224	-6.638804	839.3794954
120	20	1.42857E-06	0.08	4.767313	-0.004018	-6.594598	581.5391731

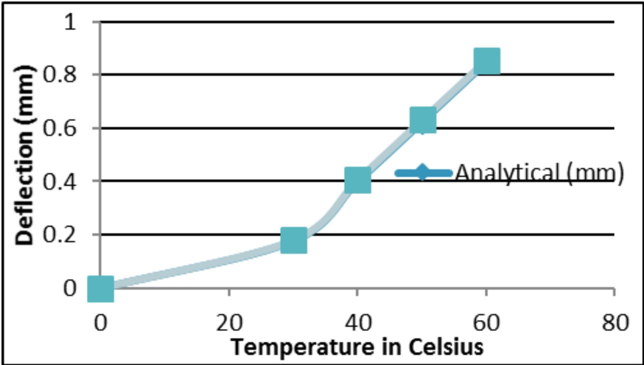
Table 3: Hydraulic diameter calculations for various IT power dissipation ratings

Here, rate of heat transfers from chips-heatsink-water, heat sink efficiency and bypass factors were neglected as the dynamic cold plate is still in design optimization stage and net values of these parameters are not available. As these parameters will be added to the calculation, changes will have to be made in pipe size diameter and flow velocities to dissipate the same heat load.

As per present calculations approx. 120W of heat load can be dissipated by completely opening the valve in a 5 mm hydraulic diameter pipe. Hence the material should have the capability to deflect by approx. 4.6 mm within a set temperature.

Chapter 5 Design Approach for a New FCD

5.1 Shape Memory Materials Vs Bimetallic Materials



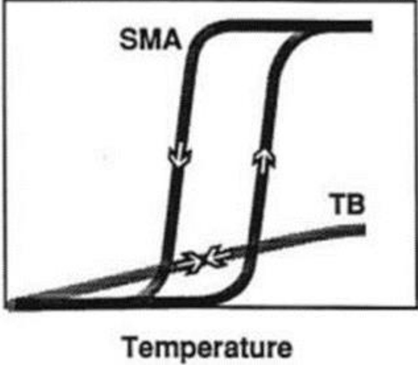


*Figure 12: Dimensions of test sample: 16*3*0.1 mm^3*[10]

Figure 13: Comparison of Thermostatic Bimetals and Shape Characteristics [12]

When Preliminary Simulations and tests were dome with the bimetallic materials for the cantilever mechanism, the deflection for the given temperature range was insignificant. Hence we needed a material that can change significantly with small change in temperature.

Looking at these results and assumptions there is a need for a material that has more deflection. From the comparison chart it is evident that shape memory materials have huge deflection for small temperature changes. Moreover, in electronics there are serious packaging challenges related to available space for different components. Nitinol has:

- Much more deflection than bimetals
- High amount of force generated when it changes shape

- Orthotropic coefficient of expansion – free of clearance issues at different temperatures unlike bimetals
- Higher elastic limit – Steel has it around 0.3% whereas nitinol has it around 8%
- Large movement w.r.t. temperature change
- Occupies much less space to do the same amount of work

5.2 Nitinol Vs Other Materials

The scale of 1-5 was assigned for the following table after referring numerous research papers, contacting various inventors and doing simulations and experimental tests on bimetallic and nitinol materials.

Materials and Selection Criteria	TM2	TM5	Treated Shape Memory Polymers	NiTi and Fluro-Polymer Fused Plate	Nitinol @ 65	Soft Polymer coating over deflecting material	NiTi @ standard and body temperature	R-Phase NiTi @ standard and body temperature	J7 - Phos Bronze Coating benefits
Composition of the Material	53% MnCuNi + 43% NiFe	80% MnCuNi + 20% NiFe	Polycaprolactones, Poltlactides, Polyester uerthane introduced with low molar mass crystallizable segments	NiTi + PEEK or NiTi + PES	NiTiCu	NiTi Coated with PES or PEEK	NiTi	NiTi	55% 925 - 5Sn,0.3Zn,Bal. Cu(C51000) + 45% 17 - Cr.Bal.Fe
Deflection	1	1	5	3	4	4	4	3	
Cost	5	5	1	4	4	2	4	2	
Corrosion	2	3	5	3	3	5	4	4	5
Energy consumed for deflection	0	0	5	1	0	3	5	5	
The amount of energy that gets transferred to water	2	2	5	3	2	5	5	5	
Relaxation time	4	4	0	4	5	2	3	3	
Repeatability of N number of cycles	4	4	0	3	5	2	5	5	
Ease of developing and availability in market	5	5	1	3	4	3	4	3	
Flexibility in operation as per aging and changing need	3	3	2	2	3	3	2	2	
Can thermal gradient within the pipe create issues	5	5	3	3	5	5	3	3	
Effect of Cooling and Heating Hysterisis curve on performance	3	3	1	3	3	3	2	3	
Delta T required for Deflection			25°C				15°C	3° to 7°C	
Total	34	35	28	32	38	37	41	38	5
Can it be used passively for given application	No	No	Yes	Yes	No	No	Yes	Yes	

Fail Criteria	Worse	Bad	Good	Excellent
0	1	2	4	5

Table 4: Study of material properties for consideration as FCD material

TM2 and TM5 materials are the thermostat materials stated in the handbook of Texas Instruments for selecting Bimetallic Materials. Present shape memory polymers are very slow reacting and not repeatable after 15 / 20 thermal cycles [18]. Moreover, all other shape memory metals i.e.- Cu-Zn-Al, Cu-Ni-Al and CU-Zn-GA do not have the necessary reliability to be used as a flow control device [11].

One of the design approach is to partially cover the deflecting material (Selecting nitinol that activates at 45C or 55C for a closed loop system where the coolant temperatures are not more than 35C.) with an insulating (elastomeric, polymeric, rubber) material and have active control. By doing this we are controlling the amount of heat that is being passed on to the coolant and at the same time have a very responsive control – which can also can be regulated as the materials sensitivity changes.

The other approach involves using the deflecting material passively – keeping it directly in contact with water.

Currently we have shortlisted following grades of Nitinol for our flow control device:

Activates at 35, 45 and 55 degrees Celsius

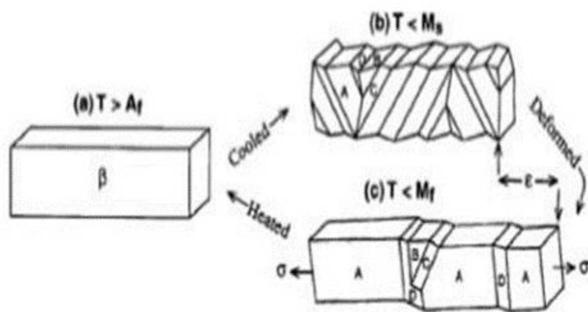
R phase nitinol at 25 ˚C, 35 ˚C and 45 ˚Celsius

Two Way Shape Memory Effect Nitinol

5.3 Nitinol (FLEXINOL®) Alloy Physical Properties [13]

Composition	Ti = 40% - 50% , Ni = 40% - 50%, Copper 3% - 5%
Density	0.235 lb/in³
Specific Heat	0.20 BTU/lb * °F (0.2 cal/g * °C))
Melting Point	2370 °F (1300 °C)
Latent Heat of Transformation	10.4 BTU/lb (5.78 cal/g)
Thermal Conductivity	10.4 BTU/hr * ft * °F (0.18 W/cm * °C)
Thermal Expansion Coefficient	Martensite 3.67x10-6 /°F (6.6x10-6 /°C) Austenite 6.11x10-6 /°F (11.0x 10-6 /°C)
Poisson Ratio	0.33
Electrical Resistivity (approx.)	Martensite: 32 micro-ohms * in (80 micro-ohms * cm) Austenite: 39 micro-ohms * in (100 micro-ohms * cm)

Table 5: Properties of Nitinol procured from Dynalloy Inc.



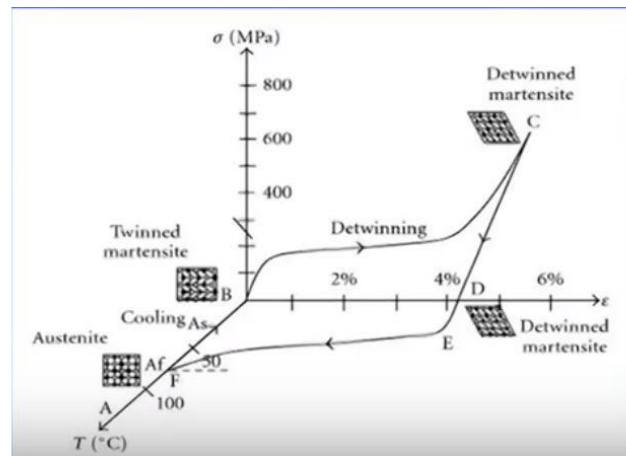


Figure 14 (left): Nitinol's Phase Transformation Cycle for a specific load [14].

Figure 15(right): Effects of stress and temperature on Nitinol [14].

For the R-phased Nitinol the hysteresis curve will be narrow than the one shown over here.

5.4 Mass of FCD Damper

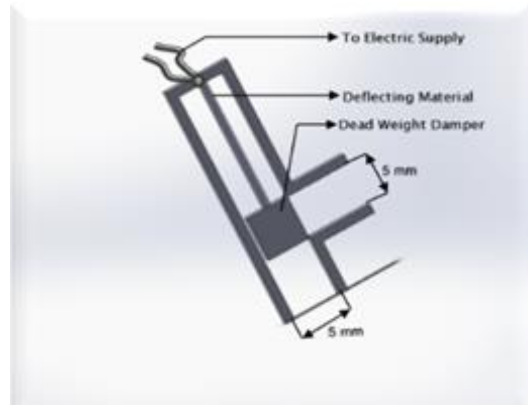


Figure 16: Typical orientation of Damper to restrict the flow of water.

This design of damper acts under 2 opposing forces – 1st the gravity in downwards and 2nd the varying spring force in upward direction.

Force on damper for 5mm Diameter pipe:

= ((Static Pressure + Dynamic Pressure) * Cross Sectional Area) + Spring force for 5mm

= (1360+((1/2) *Density*Velocity^2) *(0.000019635) + spring constant when cold * displacement

= (1360 + 19.914) * (0.000019635) + (5.5 * 0.005) = 0.0270945N + 0.0275N

Mass of the Damper that is placed on top of the flowing liquid:

Mass = (Force / Gravity) = 0.0545/ 9.8066 = 0.00555 kg= 5.55 g ~ 6 grams

This is the calculation when the damper is kept perpendicularly - if the orientation of the damper changes there may be slight c hange to the calculated value. One of the optimized design is a square cross-sectioned butterfly valve, which will have way lower mass as it will have balanced hydraulic forces on both the sides of center pivot.

5.5 Reliability and Operating Range for Nitinol

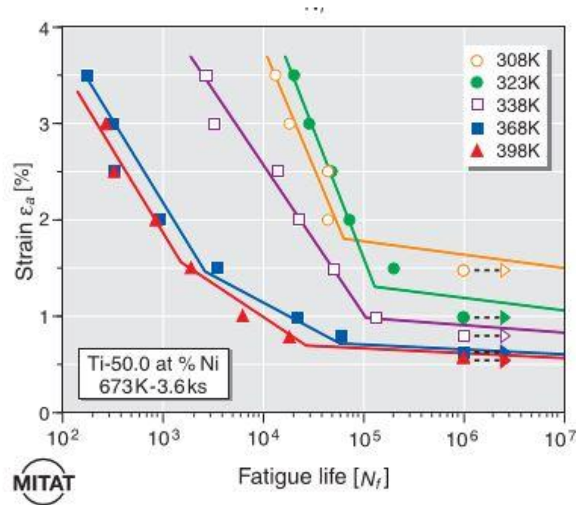


Figure 17: Graph 4: Allowable strain vs fatigue life [16]

The figure showcases the strain vs fatigue life of nitinol developed with different heat treatment. As per the various papers that were referred to use nitinol for more than 10^5 cycles the strain needs to be less than 2% and the allowable stress is less than 70 MPA.

Operating zone for nitinol wires is mentioned below for 2 %strain limit. Hence, the red zone in the graph is the region in which the material needs to be used for flow control device [13] [16].

No of Phase Change Cycles	Max Allowable % Strain[1]	Max Allowable Stress in MPa
100	4	275
10000	2	140
100000+	1	70

Table 6: Conditions for reliability of nitinol for various conditions [15]

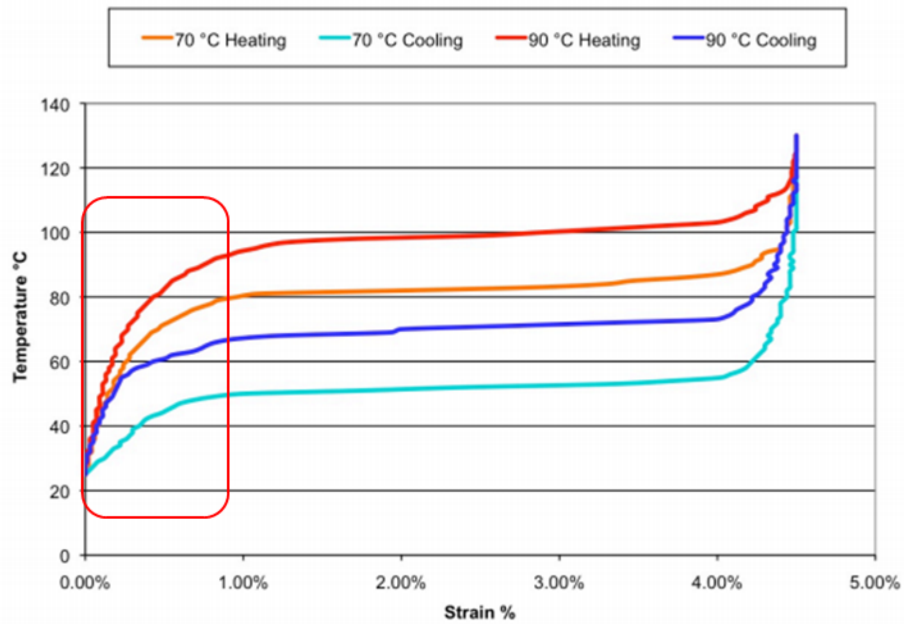


Figure 18: Hysteresis curve for wires procured from Dynalloy Inc.

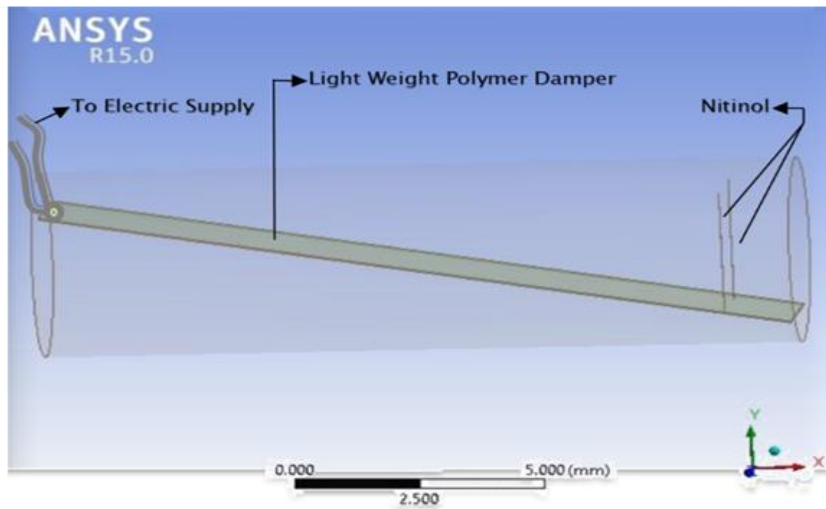


Figure 19: CAD model of wire based FCD design

Looking at the limitation of the allowable stress limits for 10^{7} cycles, the size of the FCD extended considerably. The minimum size for the FCD increases to 7cm for the wire based system and up to 3 cm for the shape changing damper based system. Also, in the cantilever beam mechanism the hydraulic forces vary as the valve opens up and the wires stiffness needs to compensate for it for precise flow control. This would create issues during pressure fluctuation in the system. Hence, there a new design approach was required which can overcome these drawbacks.

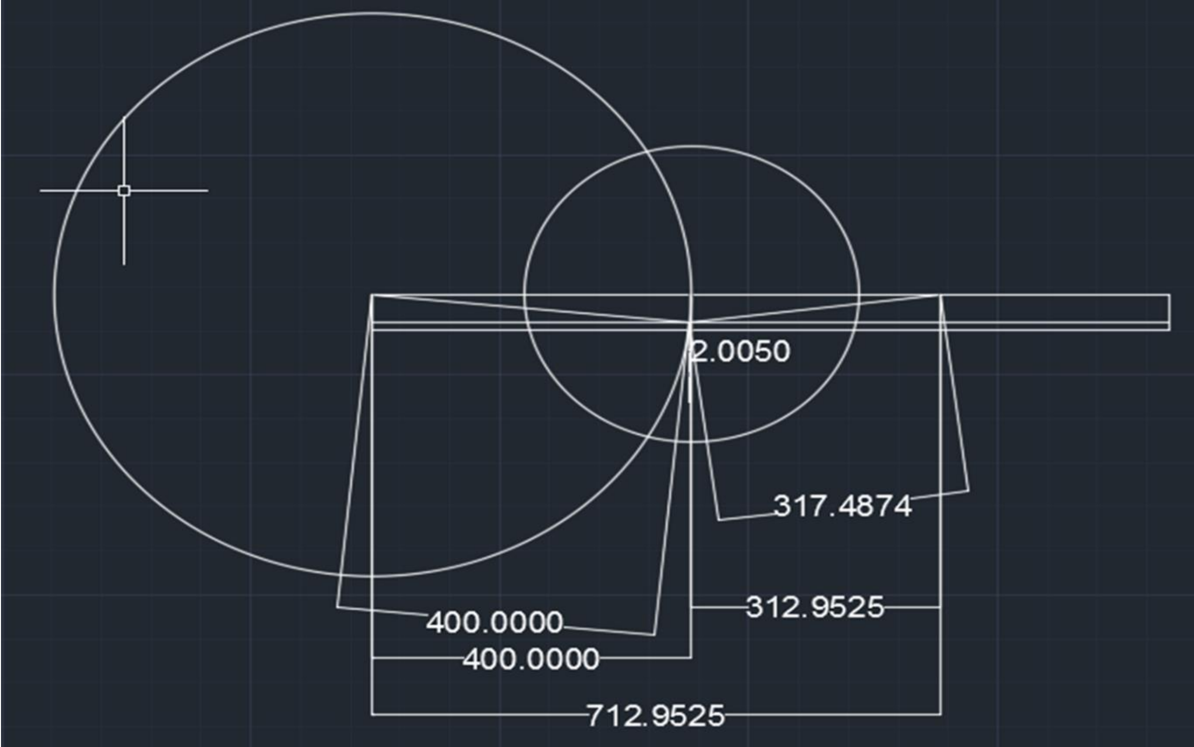


Figure 20: 2D CAD Design of the wire based FCD. Scale is 10:1.

Chapter 6 Flow Control Device Design Optimization

There were packaging issues created due to the wire based designs. The foot print of DCP is 10cm*10cm. This DCP has 4 section where in each section has footprint of 5cm*5cm. Using this design created packaging issues. Hence, we decide to not use wires or shape changing strips as the deflecting dampers but to use the springs. The advantages with these spring is that we can take advantage of change in length and stiffness to achieve the desired deflection of the FCD damper. Moreover, there were benefits observed related to springs mechanical advantage and minimum change in the design when the deflection w.r.t temperature had to be modified for customized design.

6.1 Test Results of the Temperature Sensitive Shape Changing Springs

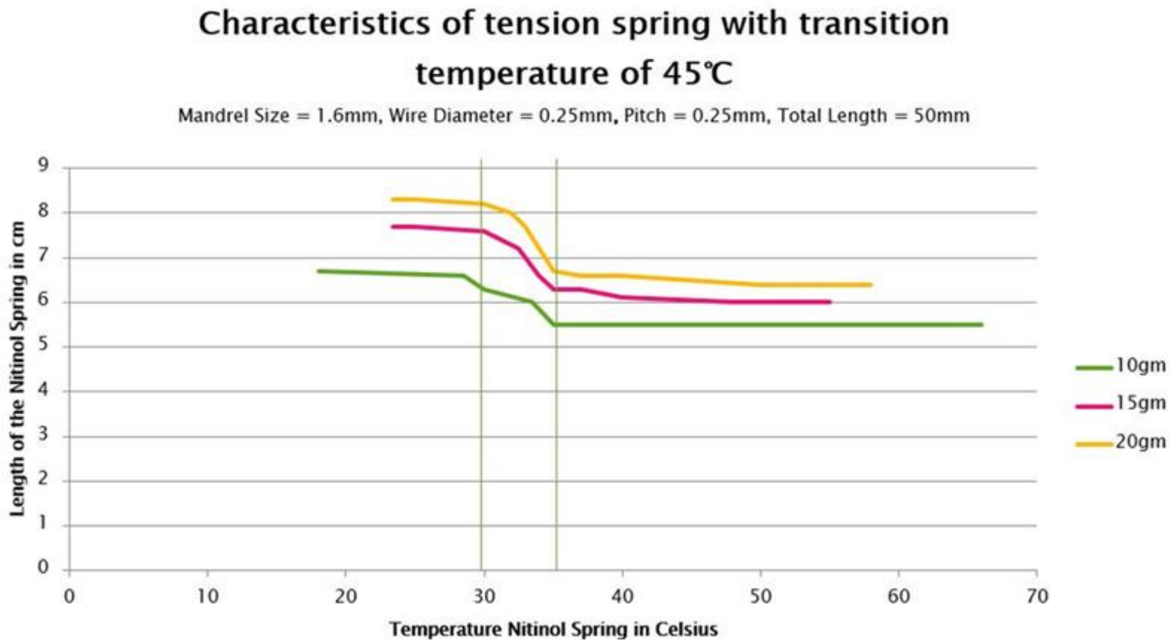


Figure 21: Length Vs temperature graph for nitinol tension spring

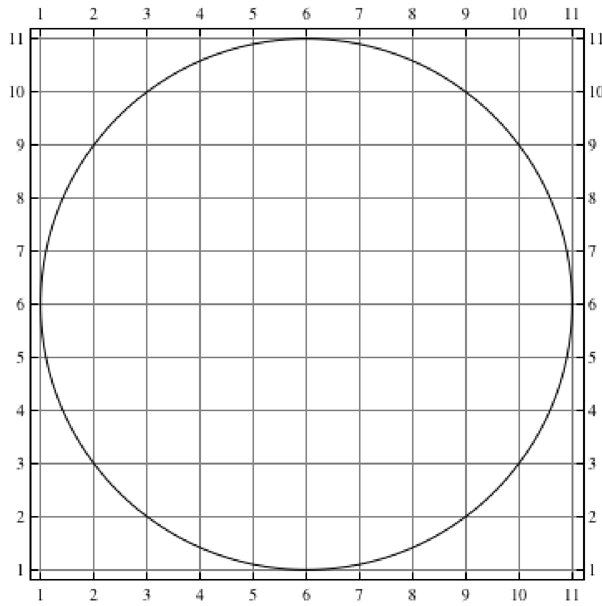
From this experiment we come to know that the material is most active in the range of 29˚C to 35˚C and has the most deflection. The deflection in the spring when 15g weight is used is approximately (7.6-6.2)/7.6 = 18% when the temperature changes from 35˚C to 29˚C. By restricting the rise in temperature we can control the max strain in the spring. The guideline would be to keep the operating temperature range of the dynamic cold plate will be between 26˚C to 33˚C when Nitinol of 45˚C is used.

Preliminary tests on torsional spring with 20 turns showed deflection in the range of 80 degrees to 120 degrees depending on the temperature achieved. More tests are under process to closely calibrate the exact range of deflection w.tr.t. temperature. Post this a prototype is under development for further testing.

Presently, tests are being conducted on various geometries and grades of nitinol to shortlist the material and manufacturer that will closely meet the narrow thermal hysteresis requirements. Also tests on bimetals are being conducted to give a comparative difference with both of these approaches i.e. using bimetals and SMA's as the deflecting material for butterfly valve mechanism.

6.2 Optimized Design

To overcome the issues related to changing instantaneous system pressures and to miniaturize the FCD a circular cross section butterfly valve design was considered. But as the butterfly valve has circular cross-section, its flow isn't linear w.r.t changing opening/ closing positions. To overcome this, a square cross-sectioned butterfly design was proposed.



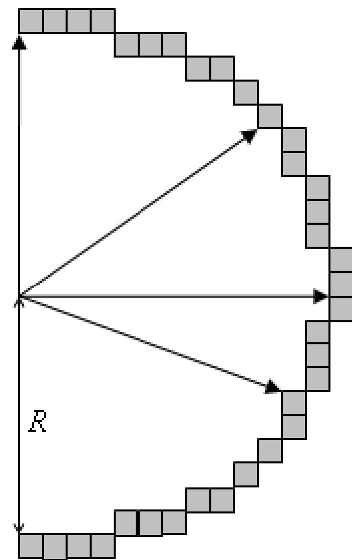


Figure 22: Front view of the circular cross-sectioned FCD.

Figure 23: Top view of a typical rotating butterfly valve.

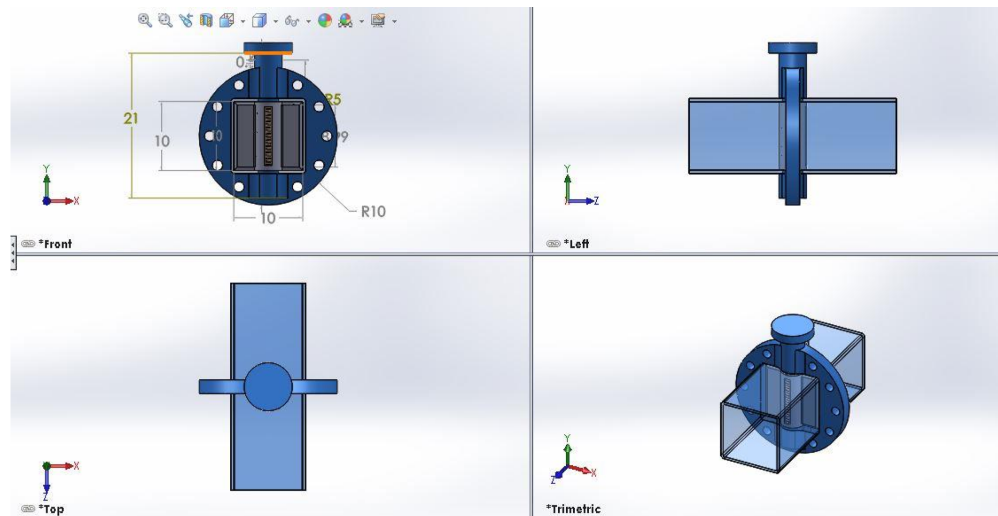
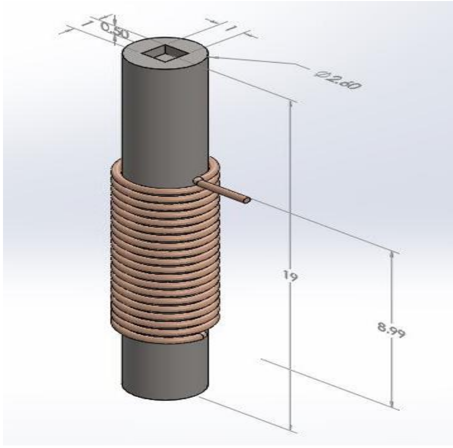


Figure 24: front view, side view, orthogonal and top view of the optimized FCD design.



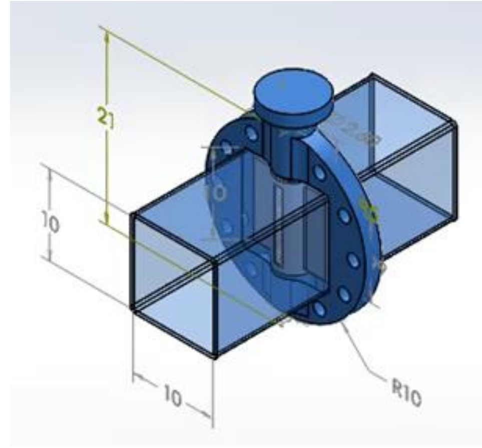


Figure 25: Temperature sensitive torsional spring

Figure 26: Orthogonal view of flow control device

Change in orientation per coil for 1% strain = 360^0 * 0.01 = 3.6^0 per turn. Total Change in Orientation of the FCD attached to the spring = 3.6^0* 25 = 90^0 rotation of valve. Hence, 90^0 rotations are achieved for a given temperature range. This FCD Designed has approx. 1% strain and 30 MPa stress (<70MPA) for 25 coil, Diameter of wire is 0.050 mm with a loading of less than 10grams. This meets the materials reliability design considerations for nitinol based systems.

The operating temperature range has to be controlled precisely to keep the FCD in best operating condition. If the temperature rises above the specified limit, then the amount of strain in the material will increase. This may lead to device failure before the predicted product life cycle.

6.3 Variations to the Design

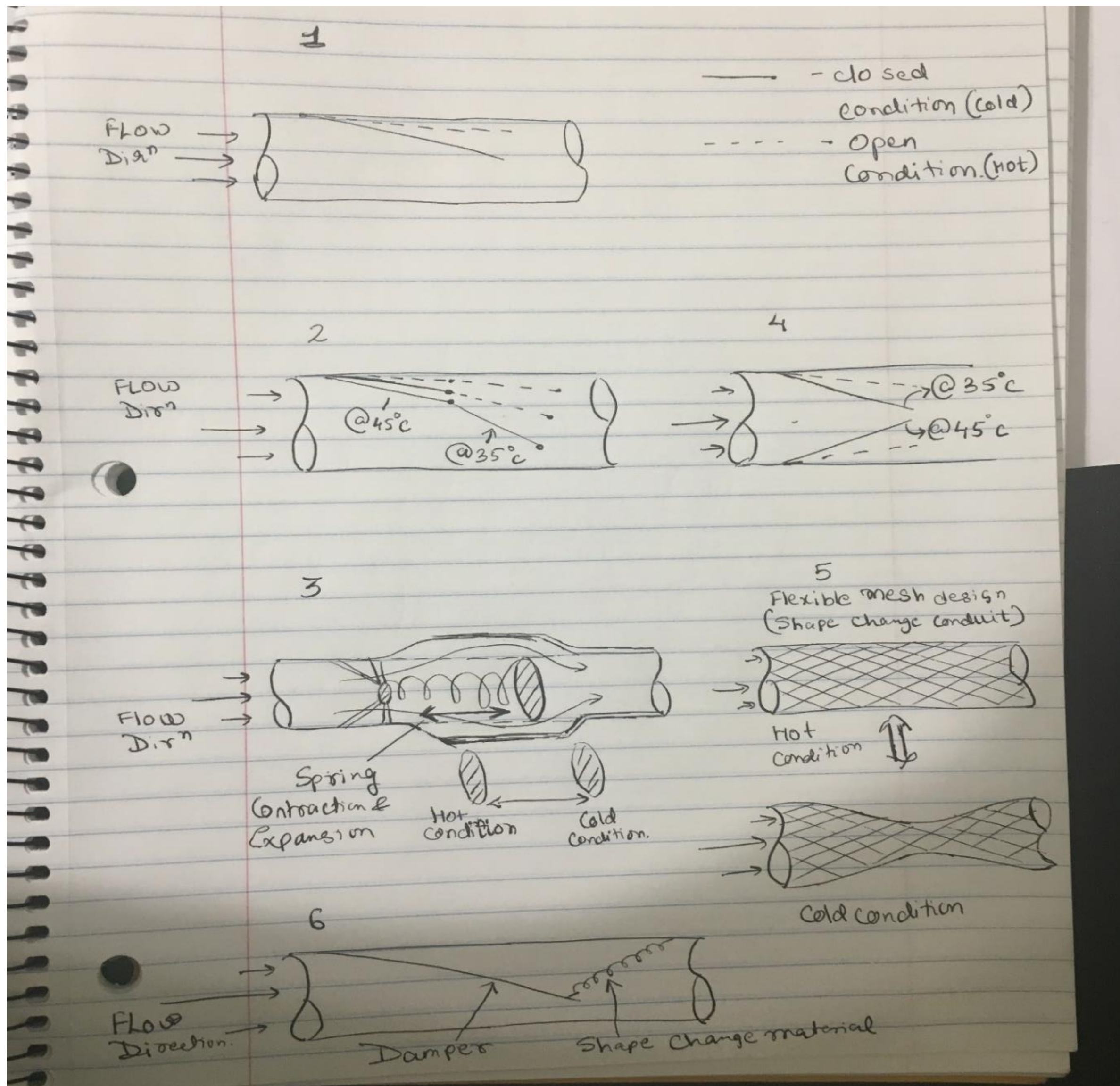


Figure 27: 6 different variations of the flow control device design

- Using Two Way Shape Memory Effect (TWME) Nitinol– No need to apply stress to bring back to its original shape
- Using trained nitinol

- R Phased Nitinol – has a thin thermal hysteresis curve up to 3 to 7 degrees
- Using different geometries other than butterfly valve to control flow – like using flexible mesh structure – shape changing tube, cantilever mechanism damper, double cantilever mechanism, double opposing loaded spring mechanism, etc.
- Using 2 or more than 2 nitinol materials that activate at different temperature to have a stepped flow regulation
- Using low cost bi-materials, bimetals instead of nitinol by increasing the number of coils on the spring, increasing the size of the valve, changing pivot diameter and hence changing on sensitivity as per system requirement.

6.4 Advantages of the New Design

- reduction in pumping power and average temperatures across the cold plate can be realized for dynamic cold plates with the help of this design
- Compact Design
- Passive Control
- Reduction in costs associated with down time, maintenance, spare inventory, data collection and performance monitoring of FCD
- Linearized flow as a result of square cross section
- Removal of fans and reduction in pumping power for the nitinol damper system
- **Lean design – A flow control device with the dimensions of 5*5*5 mm^3**

6.5 Extendibility to Other Applications

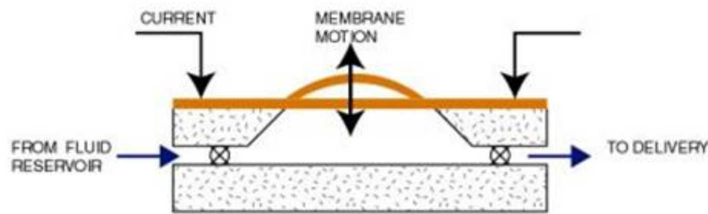


Figure 28: Nitinol pancreatic pump [17]

Figure 29: Nitinol damper [18]

- Nitinol Pumps – for laptops, tablets and mobile devices – compact design [17] (see fig. on left).
- Nitinol Dampers – Removal of Fans at each server - better control and lower pumping power [18] (see fig. on right).
- Nitinol Engine – Heat recovery system at facility level for datacenters.
- Nitinol variable heat exchanger – wherein the heat exchangers flow path is regulated by changing the orientation as the heat load changes – helps in reducing friction loss and in turn reduces pumping power.

Chapter 7 Optimization of Servers to Decrease Junction Temperature of Heat Sinks

7.1 Importance of Oil Immersion Cooling

For high end, high heat flux computing devices liquid cooling is the sought after approach because of its promise of lower PUE's and high heat carrying capacities. Now cold plate is a hybrid cooling technology, the cold plate cools the high heat flux components like GPU's and processors whereas other components are cooled by air. Moreover, water is electrically conductive and hence its leakage may lead to server failure and short circuit. Dielectric liquids like mineral oil can solve this problem. We can directly put the servers in oil baths to dissipate the heat. In this research work we are going to explore this upcoming field of oil immersion cooling.

7.2 Why Mineral Oil ?

Oil is dielectric in nature and has very high heat carrying capacity. A detailed study can for the selection of the perfect dielectric oil for datacenters needs consideration of many different parameters like reactivity with other materials and air, flammability changes in viscosity with temperature, dissociation upon thermal cycling etc. Such an analysis is out of scope for this particular study.

For this study we are considering mineral oil as our cooling medium. It was finalized after considering its extensive use in the transformer industry, low flammability compared to other petrochemical oils, low cost and simplicity in procurement.

Fluid Medium	Heat Capacity (kJ/kgK)	Conductivity (W/mK)	Thermal Mass (KJ/m³k)	Kinematic Viscosity (mm^2 /s)
Air	1.01	0.02	1.237	0.16
Dielectric Mineral Oil	1.67	0.13	1417.83	16.02
Water	4.19	0.58	4186	0.66

Table 7: Comparison of various coolant properties

7.3 Server Selection

Oil immersion seems to be a promising alternative to mitigate the challenge of increasing junction temperatures when the heat flux density increases exponentially. This is required even more when one heat generating element is placed in the shadow region of another to reduce the server's foot print. In this study, 3rd generation Open Compute Server that has a similar configuration is considered.

Figure 30: 3rd generation Open Compute server[20]

The baseline model has air pumping volumes of chilled air from one end of the server. This model is compared with mineral oil cooled system and subsequently, is compared with the geometrically optimized model for oil cooling. These models are compared at constant mass flow rate. As the oil cooling system does not require fan for displacing the media the fan components are removed and

the height of server is reduced considerably. This helps in stacking more servers within the same space.

The heat sinks geometries on both the chips are optimized to reduce the overall junction temperature. Ducting is incorporated over the RAM′s to guide majority of the flow over the heat sinks. Moreover, different server orientations are studied to see the effect of gravity on thermal shadowing and junction temperatures. All the Numerical Analysis are conducted using ANSYS Icepak software. The optimizations are conducted by using parameterization tools. The resultant effect of using mineral oil and these optimizations is that the high-end servers can have much higher heat flux density nevertheless maintain its junction temperature under acceptable limits.

7.4 Fluid Conditions

Flow Solver	ANSYS Icepak
Temperature at Inlet	30 ˚ C
Volume Flow Rate	1 lpm
Velocity	0.001960784 m/s in Y-direction
Pressure	6 psi
Reynolds Number (Re*)	64.027

Table 8: Fluid conditions for simulations

*$Re = \rho u d_h / \mu$,

$d_h = 2*g*h/ (g+h)$

Where g = channel width, h = fin height

7.5 Mineral oil and Server Material Properties

Mineral Oil:

Density - 849 Kg/m3

Specific heat -1.680 KJ/kg K

Thermal conductivity – 0.13 W/m K

$\mu = C1 \times e^{(\frac{2797.3}{T+273.2})}$

Where, μ = Dynamic viscosity (centipoise)

T = Temperature (°C)

C_1=Coefficient for scaling (Value of 0.001383 at 40°C value from STE Oil data sheets)

PCB - FR-4 epoxy:

Density- 1900 kg/m3

Thermal conductivity – 0.17 W/m K

Specific heat – 749 J/kg K

Heat sink - Aluminum:

Density- 2700 kg/m3

Thermal conductivity-218 W/m K

Specific heat- 900 J/Kg K

7.6 Server Geometry

For this study we are considering air cooled 3rd generation open compute server as the base line model. For simplicity, geometries of PCB, CPU's, DIMM's, heat sinks are the only ones which are modelled and only the region of high heat spot are powered for simulation results i.e. the CPU.

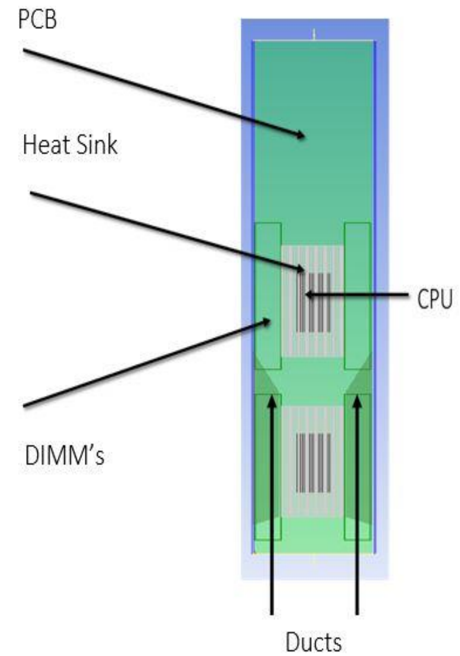


Figure 31: Server geometry

1) PCB FR-4

   Material: epoxy resin

   Foot Print: 165mm * 508mm

   Height: 2.5mm

2) Heat Sink

   Material: Aluminum

   Foot Print: 85mm * 110mm

   Base Height: 5mm, Overall Height: 43mm

   No of Fins: 34

3) DIMM's (16 Nos.)

   Foot Print: 36mm*145mm (4 dims arranged in parallel)

   Height: 33mm

4) CPU's (2 Nos.)

   Foot Print: 5cm * 5 cm

Chapter 8 Server Design Optimization Approach

8.1 Heat Sink Optimization

Heat kinks are passive heat exchangers that transfer the unwanted excess heat from the processor to the cooling medium. When kept over the processer it permits dense loading of the processor. There are various types of heat sinks available like pin fin, plate fin, staggered plate fin, parallel plate fin, heat pipes within the heat sinks, etc. For this study we are considering only the parallel plate fins. The advantages of optimizing the heat sinks are:

- Reduction in form factor
- Reduction in material wastage
- Reduction in max junction temperature
- Reduction in pumping power
- Overall increase in server efficiency and reliability

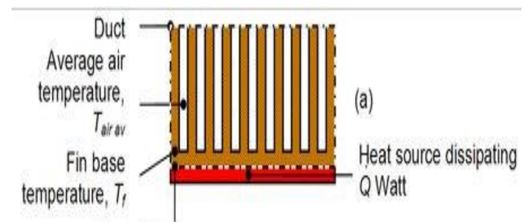


Figure 32: Heat sink design[21]

Thermal Resistance = (Tj - Ta) / Heat Dissipation from processor in Watts

Tj = Junction Temperature in Celsius, Ta = Ambient Temperature in Celsius (inlet = 30° C)

For this study, our primary approach is to decrease the max. junction temperature of the server. Hence, the thermal resistance of the CPU in the shadow region has to be reduced. Our approach is to keep the fin density of the heat sink in shadow region as 34 and the height of the fins to be 43 whereas the fin density and height of the first heat sink is optimized. As a result, the cooling fluid carries less heat from the first heat sink and more from the second heat sink. One of the purpose of this study is to determine the optimum design of the 1st heat sinks used in oil cooled 3rd generation open compute server.

8.2 Duct Design Optimization

Fluids and electrons always traverses the path of least resistance. Out of the two parallel fluid flow paths that are available fluid takes the one with lease obstructions. The height of the DIMM's is lower than the height of the heat skinks moreover there is no resistance above the fins as compares to the resistance of fins of the heat sink over the 115 W CPU's. As a result, a large quantity of the fluid by-passes over the DIMM's and the servers are left undercooled w.r.t the flow rate and heat carrying capacity of the fluid. To mitigate this issue, we incorporate ducts. Ducts are placed over the DIMM's to direct maximum amount of flow through the heat sinks. 5 different iterations of the Duct Designs were analyzed. Parameters that were changed are the length of the ducts, convergent and divergent angles, positioning of the ducts w.r.t. CPU 1 and CPU 2.

Here Design 2 and Design 5 were showing the best results as the regions of max thermal resistance i.e. the heat sink regions had maximum fluid flow.

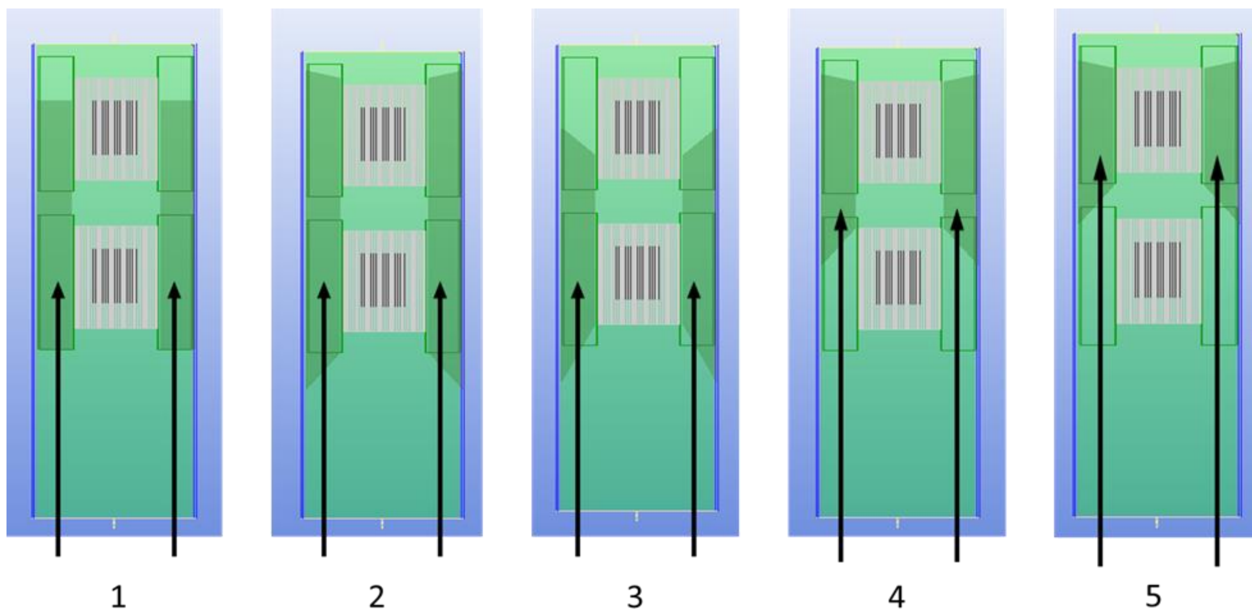


Figure 33: Duct design iterations for optimization

8.3 Server Orientation and Size Optimization

Other affecting parameters like the effect of gravity and shadow effect on the performance are also analyzed. Servers orientation at 0˚, 45˚ and 90˚ are studied and the one at 90˚ shows lower junction temperature and pumping power compared to other designs. This is because of the effect of natural convection and decreasing fluid density as it heats up.

Moreover, when this server was used for air cooling, fans had to be placed at one end to allow the air to flow when it heats up also the form factor used was 2U to accommodate the fans. For oil cooling systems fans are not required and hence the server's height can be reduced from 2U to 1U.

Chapter 9 Optimized Results

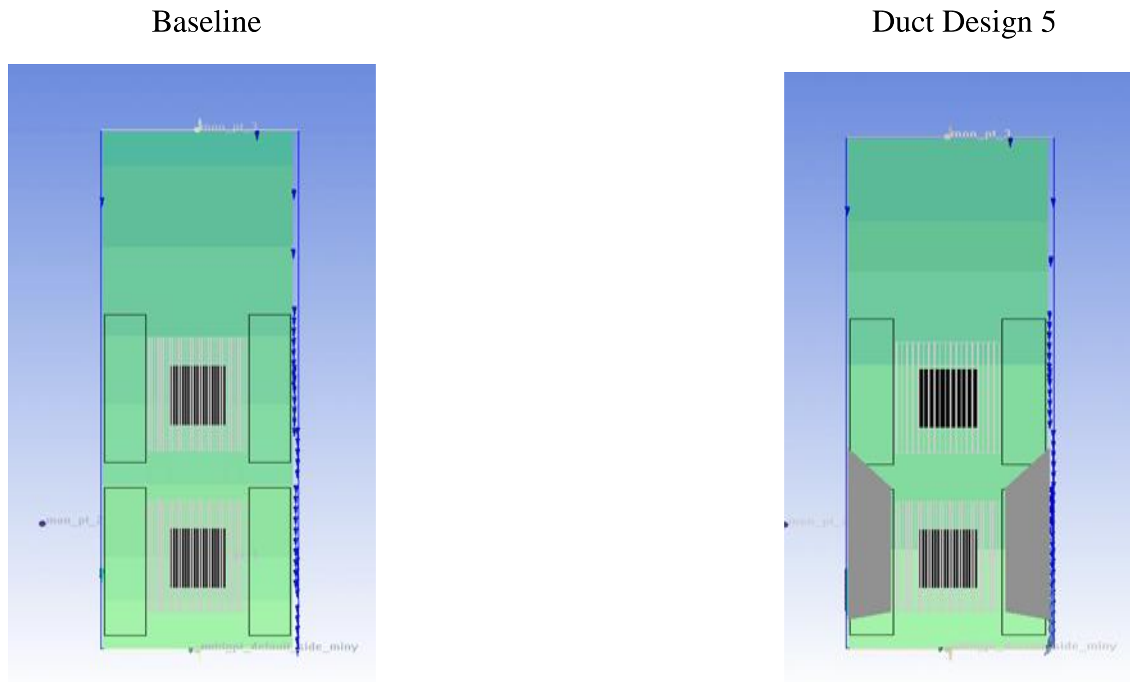


Figure 34: Comparison of baseline and optimized design

Here 2 designs are compared 1st without the duct and 2nd with the ducts and optimized heat sink 1. These simulations are carried at the same fluid conditions. Flow bypass over the DIMM's is now going through the heat sinks as the design is optimized. The flow not only increases over the heat sink in shadow region but also through the heat sink 1. This considerably reduces the thermal resistance and junction temperature on the server.

The best results are obtained when the heat sink 1 has height of the fins is 40 to 43 and no of fins are 20, Vertical orientation and duct design 5

Baseline

Duct Design 5

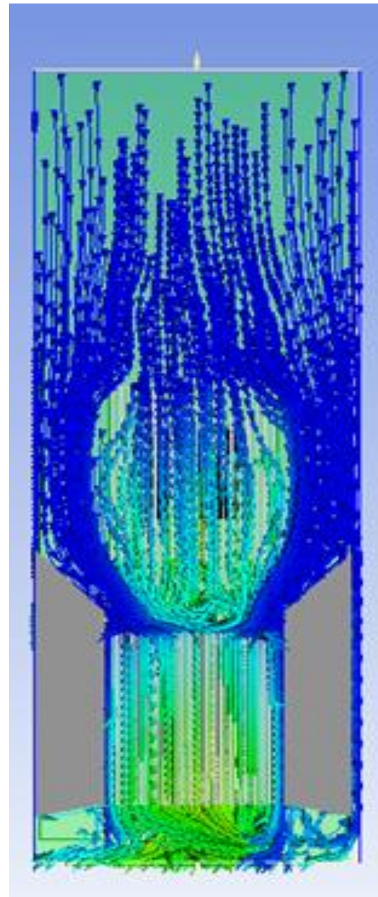


Figure 35: Comparison of streamlines for baseline and optimized servers

LIST OF FIGURES

Figure 1: Order of magnitude for heat transfer coefficients depending on cooling technology [6]. 8

Figure 2: Top view of Original Cold Plate [7] 14

Figure 3: Top view of Dynamic Cold Plate [7] 14

Figure 4: Multi Chip Module for the Cold Plate [7] 15

Figure 5:Power Rating of Chips [7] 15

Figure 6 (left): CAD model of the Dynamic Cold Plate [7] 16

Figure 7 (right): CFD model of the Dynamic Cold Plate [7] 16

Figure 8 (left): Modern Control Engineering [8] 17

Figure 9 (right): – Active flow control [9] 17

Figure 10 (left) – Passive flow control by using banana leaf bimetallic strip concept [10]........... 17

Figure 11 (right): Deflection vs temperature graph for the bimetallic strip model [10]................ 17

Figure 12: Dimensions of test sample: 16*3*0.1 mm^3[10] 21

Figure 13: Comparison of Thermostatic Bimetals and Shape Characteristics [12] 21

Figure 14 (left): Nitinol’s Phase Transformation Cycle for a specific load [14]. 24

Figure 15(right): Effects of stress and temperature on Nitinol [14]... 24

Figure 16: Typical orientation of Damper to restrict the flow of water...................................... 25

Figure 17: Graph 4: Allowable strain vs fatigue life [16] 26

Figure 18: Hysteresis curve for wires procured from Dynalloy Inc. 27

Figure 19: CAD model of wire based FCD design 27

Figure 20: 2D CAD Design of the wire based FCD. Scale is 10:1 28

Figure 21: Length Vs temperature graph for nitinol tension spring 29

Figure 22: Front view of the circular cross-sectioned FCD. 31

Figure 23: Top view of a typical rotating butterfly valve. 31

Figure 24: front view, side view, orthogonal and top view of the optimized FCD design. 31

Figure 25: Temperature sensitive torsional spring 32

Figure 26: Orthogonal view of flow control device 32

Figure 27: 6 different variations of the flow control device design 33

Figure 28: Nitinol pancreatic pump [17] 35

Figure 29: Nitinol damper [18] 35

Figure 30: 3rd generation Open Compute server 37

Figure 33: Duct design iterations for optimization 42

Figure 34: Comparison of baseline and optimized design 44

Figure 35: Comparison of streamlines for baseline and optimized servers 45

LIST OF TABLES

Table Page

Table 1: Characteristics of the coolant fluid………………………………………… 8

Table 2: Power calculation to raise the FCD temperature to 70˚C Celsius……...18

Table 3: Hydraulic diameter calculations for various IT power dissipation…….20

Table 4: Study of material properties for consideration as FCD material……...22

Table 5: Properties of Nitinol procured from Dynalloy Inc……………………….25

Table 6: Conditions for reliability of nitinol for various conditions [15]………...26

Table 7: Comparison of various coolant properties……………………………...37

Table 8: Fluid conditions for simulations…………………………………….......38

LIST OF REFERENCES

[1] HTTPS://WWW.ELECTRONICS-COOLING.COM/2005/05/REVIEW-OF-LOW-PROFILE-COLD-PLATE-TECHNOLOGY-FOR-HIGH-DENSITY-SERVERS/

[2] HTTP://WWW.AAVID.COM/SITES/DEFAULT/FILES/LITERATURE/VORTEX-LIQUID-COOLING.PDF

[3] THESIS - EXPERIMENTAL ANALYSIS VALIDATING A CONTROL SCHEME TO DEVELOP A DYNAMIC COOLING SOLUTION FOR NON-UNIFORM HIGH POWERED ELECTRONIC DEVICES IN DATA CENTER, RUTURAJ KOKATE, UNIVERSITY OF TEXAS AT ARLINGTON

[4] HTTP://WWW.DATACENTERDYNAMICS.COM/CONTENT-TRACKS/DESIGN-BUILD/GLOBAL-FORECAST-BRIGHT-FOR-THE-DATA-CENTER-CONSTRUCTION-MARKET/93014.FULLARTICLE

[5] HTTP://WWW.INFOWORLD.COM/ARTICLE/2895067/DATA-CENTER/FACEBOOK-OPEN-COMPUTE-PROJECT-BILLIONS-IN-SAVINGS.HTML

[6] HTTP://WWW.ELECTRONICS-COOLING.COM/2005/11/ADVANCES-IN-HIGH-PERFORMANCE-COOLING-FOR-ELECTRONICS/

[7] FIGURE 2, 3, 4: THESIS - PERFORMANCE EVALUATION OF PLATE-FIN AND PIN-FIN HEAT SINKS FOR THE APPLICATION OF OIL IMMERSION COOLING AND DESIGN OPTIMIZATION OF DYNAMIC COLD PLATE FOR THE APPLICATION OF WARM WATER COOLING, PARTH SONI, UNIVERSITY OF TEXAS AT ARLINGTON

[8] FIGURE 5: MODERN CONTROL ENGINEERING, 5TH EDITION – KATSUHIKO OGATA

[9] FIGURE 6: HTTP://WWW.FLOWSERVE.COM/PRODUCTS/VALVES/CONTROL SOURCE PROJECT 6 ES2 CONFERENCE, JOHN FERNANDEZ, UNIVERSITY OF TEXAS AT ARLINGTON

[10] FIGURE 7, GRAPH 1: PASSIVE FLOW CONTROL: THESIS - CONTROLLING FLOW USING BIMETALLIC STRIPS, RAVI TEJA MUTULAYA, UNIVERSITY OF TEXAS AT ARLINGTON.

[11] HTTPS://WWW.YOUTUBE.COM/WATCH?V=YJ0WWW0AOUK – NCTEL LECTURE

[12] THERMAL ACTUATION WITH SHAPE MEMORY ALLOYS - DIETER STOECKEL, RAYCHEM CORPORATION (WWW.NITINOL.COM)

[13] HTTP://WWW.DYNALLOY.COM/PDFS/TCF1140_2.PDF

[14] PHASE CHANGE BEHAVIOR OF SHAPE MEMORY ALLOYS, BY JOSE MARIA GALLARDO FUENTES, PAUL GUMPEL, JOACHIM STRITTMATTER, ADVANCED ENGINEERING MATERIALS

[15] HTTPS://WWW.NITINOL.COM/WP-CONTENT/UPLOADS/2012/01/048.PDF

[16] STRAIN VS FATIGUE LIFE - OPTIMISATION OF PROCESSING AND PROPERTIES OF MEDICAL GRADE NITINOL WIRE, A.R. PELTON, J. DICELLO AND S. MIYAZAKI

[17] HTTP://MEMRY.COM/SITES/DEFAULT/FILES/DOCUMENTS/ARTIFICIAL_PANCREAS_NITINOL_PUMP_SMST03.PDF

[18] SHAPE ACTUATION VIA INTERNAL STRESS-INDUCED CRYSTALLIZATION OF DUALCURE NETWORKS - YUAN MENG, JISU JIANG, AND MITCHELL ANTHAMATTEN – ACS MACRO LETTER, 2015

[19] RECEIVED BEST STUDENT ABSTRACT AWARD WHEN IT WAS PRESENTED AS 'DESIGN OF AN IMPROVED FLOW CONTROL DEVICE FOR DYNAMIC COLD PLATES TO HAVE

distributed cooling in a water cooled multi chip module' as a part of the 'es2 project 6 – design of dynamic cold plates' at imaps atw and tabletop exhibits on thermal management, toll house, october 25-27, 2016, los gatos, california

[20] Figure 30: http://www.opencompute.org/blog/author

[21] Reference 1: Kordyban, T., 1998, Hot air rises and heat sinks – Everything you know about cooling electronics is wrong, ASME Press NY

BIOGRAPHICAL INFORMATION

Kunal Atulkumar Shah was born in Mumbai, India in December 1990. He completed his Bachelors of Engineering in Mechanical Engineering from A. D. Patel Institute of Technology (Gujarat Technological University) in May 2012. Post which he went for industry experience for 2 years. Later on he joined University of Texas at Arlington to pursue Masters of Science in Mechanical Engineering. He completed his thesis work under the guidance of Dr. Dereje Agonafer in December 2016.

During his time at U.T. Arlington he worked as a Graduate Research Assistant at University's patent office (Office of Technology Management) and also worked on various thermal projects like designing cooling architecture for electric vehicles to increase heat exchanger efficiency and reduce system weight, designing a 5*5*5 mm^3 flow control device for dynamic cold plates and optimizing 3rd generation Open Compute servers by reducing the bypass factor considerably. All these projects are relevant to the upcoming technologies in the automotive and electronic packaging domains.

At University of Texas at Arlington he was honored with Graduate Research Assistantship by Office of V.P for Research and Graduate Scholarship by The Electronic, MEMS and Nanoelectronics Systems Packaging Center (EMNSPC). He received Best Student Abstract Award at International Microelectronics Assembly and Packaging Society - ATW 2016 for his work of designing flow control device for dynamic cold plates.

His industry experience includes working with firms like Ghafari Associates to design utilities for Ford's engine plant to Bosch Automotive Service Solution for testing connected cars. He has served in job roles ranging from Mechanical Design Engineer, Marketing / Technology Analyst to Test Engineer / Strategic Marketing Intern.